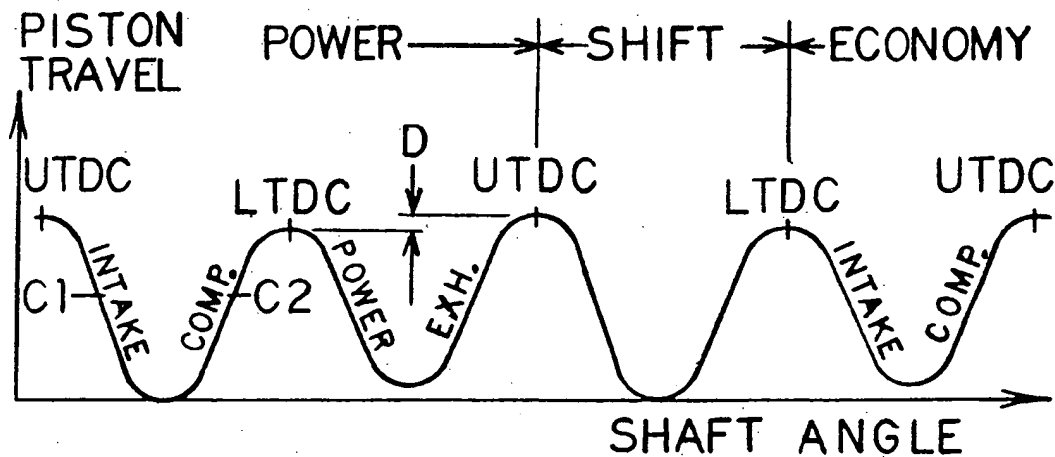




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(54) Title: A DUAL MODE, PHASE SHIFTING, CAM ENGINE



(57) Abstract

A combination, in an expansible chamber engine having at least one cam driven piston (174), of a piston drive cam (18) profile that alternately drives the piston to a higher or lower top dead center TDC position producing different expansion ratios, of a valve operating arrangement that shifts firing position between the two TDC positions selecting an expansion ratio, of a charge volume limiting system that limits the charge by controlling intake valve open duration and reduces throttling losses, and of a control system that limits the maximum charge volume or intake displacement in accordance with the firing TDC and the supercharged pressure thereby avoiding pre-ignition firing and allowing supercharger compression to replace cylinder compression instead of adding to it.

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A DUAL MODE, PHASE SHIFTING, CAM ENGINE

Background-Field of Invention

This invention relates to an expansible chamber engine with cam driven pistons, that during operation can change expansion ratios and also intake displacements to limit the fuel air charge. Specifically to an arrangement that shifts combustion peaks between, differing in height, top dead center positions on a four stroke piston drive cam, by shifting valve and combustion timing and limiting the charge to prevent pre-ignition firing caused by the shift, by controlling open duration of the intake valve, and/or limiting maximum charge volume so the work used to supercharge replaces cylinder compression.

Background-Description of Prior Art

Increasing fuel efficiency or decreasing specific fuel consumption, SFC, in commercially acceptable engines has been restricted in many ways:

(a) Limiting is defined for this invention as the process of controlling the unthrottled fuel air charge into a cylinder by closing the intake valve early. In a limited engine as in U.S. Pat. 4,280,451, the compression ratio was increased by shaving the heads. The cylinder intake volume was reduced by limiting. The resulting compression ratio, herein called the limited compression ratio, is limited by pre-ignition firing to the same maximum. But, the volumetric efficiency is decreased requiring a larger engine for the same power.

(b) Decreasing SFC by increasing expansion ratio is restricted by current designs offering only fixed and equal expansion and compression ratios.

(c) Recent cylinder cutout systems are slow multi-revolution devices, not capable of single revolution control.

(d) Decreasing SFC by increasing the compression ratio, or more relevantly the pre-ignition pressure to atmospheric pressure ratio, is restricted by the maximum pre-ignition pressure and temperature that avoids pre-ignition firing.

(e) There have been recent attempts to decrease SFC by reducing displacement during operation. Deactivating valves, switching off cylinders, and unfortunately cooling cylinders. So far not commercially practical.

(f) Automobiles mostly operate at 10 to 20 percent of maximum power output. Unfortunately efficiency deteriorates with reductions in load, thus engines operate in their worst SFC region.

(g) Controlling engines by limiting produces a higher hydrocarbon content in the waste gas than does throttling. Specifically in the idle and lower partial load regions, as discussed in U.S. Pat. 4,765,288. Briefly, after valve closure, the charge expands to fill the full cylinder and then is recompressed. This expansion cools the charge mixture, albeit only momentarily, until the charge is recompressed to the original volume that occurred when the valve closed. The theory stated is that "the fuel cools relatively too much, the fuel evaporates poorly and as a result poor mixture preparation takes place" causing the higher hydrocarbon level.

(h) One method for controlling valve closure hydraulically is disclosed in U.S. Pat. 4,466,398. Valve operation occurs as a translating fluid plug, interposed between a camshaft and a valve, is collapsed and refilled. This requires a hydraulic system with the capacity to rapidly refill the fluid plug, an electronic system to sense, compute, amplify and send a signal to release the fluid each cycle, and a fluid releasing valve. A complex and costly system.

(i) Supercharging is used to produce more power from an engine. Any lowering of SFC is due to increasing the mechanical efficiency, not to recovering additional work. Supercharger compression adds to cylinder compression. The total is limited by pre-ignition firing. In the crossover speed ranges where the cylinder fills without supercharging and yet there is substantial supercharging, the cylinder compression ratio is lowered to avoid pre-ignition firing. At lower speeds with little or no supercharging, the total compression ratio is the lowered cylinder ratio, increasing the SFC. Compensating for supercharger output by early intake valve closure was disclosed in Deutsche Patentschrift DT-PS 100 1049. Adaptable to large

steady state diesels and gas engines, such as 2500 horsepower, it is too complex for automotive use.

(j) In throttled engines the high vacuum that occurs during deceleration causes rapid evaporation of liquid fuel from the intake manifold walls, increasing exhaust emissions of carbon monoxide CO and hydrocarbons HC.

(k) The high temperatures during the combustion process produces nitrous oxides NOx. Catalytic reactors must be used to reduce these emissions.

(l) Maximum torque occurs when cylinders are fully charged or loaded at maximum compression ratio. Supercharging increases the mechanical efficiency and relatively lowers the expansion ratio, lowering efficiency. It, does not effectively increase the output per unit displacement, since the displacement is increased by adding the supercharger.

Objects and Advantages

Accordingly, several objects and advantages of the present invention follow in respective order and more will be apparent in the following sections:

(a) To operate an engine selectably at maximum volumetric efficiency, or shift, to a more fuel efficient cycle.

(b) To decrease SFC by utilizing a greater expansion ratio cycle.

(c) To enable orchestrating valve and cylinder operation for each revolution and thus enable the rolling deactivation of cylinders called "skipfire", etc..

(d) To decrease SFC by increasing the pre-ignition pressure at part load.

(e) To reduce engine displacement without shutting off cylinders.

(f) To improve part load SFC over full load.

(g) To improve the pre-ignition conditions when limiting to substantially reduce the higher hydrocarbon level.

(h) To provide a practical hydro-mechanical valve with reduced hydraulic flow rate that needs only passive control for steady state operation.

(i) To enable the feedback of supercharger compression work into the engine.

(j) To reduce the CO and HC emissions from rich mixture produced by high manifold vacuum during deceleration and idle.

(k) To reduce NOx emissions from the engine cylinder.

(l) To increase the supercharged output per unit of true displacement.

Drawing Figures

Fig. 1 is a simplified sectional view of of a dual compression ratio

engine.

Fig. 2 is a graph of strokes and piston travel vs. shaft angle.

Fig. 3 is a graph of: pre-ignition pressure, P_{pi} ; temperature, T_{pi} ; pressure ratio, R_{pi} ; indicated thermal efficiency, ITE ; all with respect to the percentage of maximum indicated mean effective pressure, $ZIMEP$.

Fig. 4 is a fragmentary cross-sectional view of a differential style phase shifting, and continuously variable limiting system.

Fig. 5 graphs cam follower excursion vs. main shaft angle for Figs. 4 & 8.

Fig. 6 is section of a phase shifting system with a dual lobe valve cam.

Fig. 7 graphs cam follower excursion vs. shaft angle, for Fig. 6.

Fig. 8 is the same as Fig. 4, except showing an incrementally variable limiting and a trip type phase shifting systems.

Fig. 9 is a sectional view A-A from Fig. 8, showing the trip mechanism.

Fig. 10 is a schematic representation of a control system for fig. 1 & 4.

Description/ Operation, dual compression ratio, Fig. 1

Description: A double sided piston drive cam 18 has a cam shape that undulates over the outer diameter of a drum shaped section of a main drive shaft 54. Cam 18, the drum section and shaft 54 form a rotor which is rotatably mounted in a cylinder block assembly 176. A number of rollerized double ended pistons 174 are spaced around the circumference of the rotor and each piston end is slidably engaged within the respective cylinders. The rollers are rotatably mounted in piston 174 and rollably engaged with cam 18 such that the axial position of piston 174 is determined by cam 18. Cam 18 has two maximal positions in each axial direction that differ by a dimension D.

Operation: Pistons 174 drive and are driven back and forth by rotating cam 18, in the manner of IC cam engines. The difference being that the two maximal positions, respective to each piston end on cam 18, produce two different piston up positions. An upper top dead center position called UTDC and a lower top dead center position called LTDC. The rotation of shaft 54 thereby produces a periodic succession of clearance volumes for each piston end. The smaller clearance volume 178 at UTDC produces a higher compression ratio. The larger clearance volume 172 at LTDC produces a lower compression ratio.

The periodic succession of maximal and minimal chamber volumes can be

seen in Fig. 2, wherein the strokes of a four stroke engine, intake, compression, power or expansion, and exhaust, are designated on the piston travel predetermined by cam 18. The strokes are divided into a power section where the engine operates similar to prior art fashion, an economy section where the engine operates more efficiently at the higher compression ratio, and a shift section illustrating the two stroke or 180 degree phase shift required to go from one to the other, in a manner to be explained later.

Before the phase shift, power operations: Ignition occurs at LTDC, at the relatively lower compression ratio and at the larger clearance volume. The after-exhaust clearance volume is then the smaller clearance volume, yielding a smaller residual gas fraction. The compression and expansion ratios are equal. Unlimited filling of the cylinder with fuel air charge is permitted. Under these conditions the engine can produce maximum power. They are also the conditions at which the maximum compression ratio appropriate to design considerations is set.

During the phase shift: The relationship of piston travel to strokes is shifted two strokes. This corresponds to 180 degrees of main drive shaft rotation for a four stroke piston drive cam. The ignition or combustion point is shifted from LTDC to UTDC, or the reverse when shifting the opposite direction. Valve operation may be deactivated during the phase shift, depending on considerations such as valve to piston interference, backfire, etc.. More sophisticated systems could close the valves in each cylinder late in the exhaust stroke and re-activate operation during the new exhaust stroke, for a smoother shift.

After the phase shift, economy operations: Ignition occurs at UTDC, at a higher compression ratio and at the smaller clearance volume. The after exhaust clearance volume is the larger clearance volume, yielding a larger residual gas fraction. But, the maximum compression ratio was set for conditions before the shift, with the cylinder operating in power mode. This requires that the maximum charge be limited, either by throttling to limit the charge density, or limiting to limit the charge volume by early or late intake valve closure, or both. By throttling or varying the charge density, wide open throttle position is throttled back, the throttle opening reduced, so as to maintain a maximum intake manifold pressure. By limiting or varying the charge volume, the intake valve is closed either earlier or later to limit back the maximum intake volume. By both, each would be limited in accordance with the other and the total required. The net result of economy mode is the expansion ratio is increased to the

total cylinder volume divided by the smaller clearance volume. And, the intake displacement is reduced, reducing the charge and the output of power. Further, the limited compression ratio is maintained at the reduced displacement.

An advantage of maintaining the limited compression ratio is that the SFC is decreased by increasing the pre-ignition pressure, or the pre-ignition pressure ratio with respect to atmospheric pressure, for part load operation. The pre-ignition pressure, P_{pi} in psia, shown on the left ordinate in Fig. 3, is plotted with respect to output shown on the abscissa. The plot also reflects the pressure ratio of pre-ignition to atmospheric pressure, R_{pi} , shown on the right ordinate. The output is expressed as the percentage of maximum indicated mean effective pressure, $\% imep$. It can be seen that the invention pressure ratio is considerably higher than either the throttled or the limited ratio. This contributes to the similarly higher plot of indicated thermal efficiency, $\% ITE$, shown for the same conditions. The curves of Fig. 3 are plotted from calculations of the thermodynamic conditions for various operating modes in spark ignition engines. They are based on a compression ratio of 8.9 and an expansion ratio of 15. They are for idealized operation and have not been modified to include losses. As such, they are valid only for relative comparison.

A further advantage is apparent for part load operation when referring to Fig. 3. At 59 percent IMEP, this invention results in an increase in $\% ITE$ of 21 percent. From 42 percent $\% ITE$, for an approximately equivalent IMEP under prior art or power operation, to 51 percent $\% ITE$ for economy operation. The $\% ITE$ of 51 percent is not only improved over the 42 percent at the equivalent output for power operation, it exceeds the $\% ITE$ of 46 percent at full load. In other words, this invention engine at part load is more fuel efficient than at full load. The reverse of prior art.

Description/Operation, Fig. 4

Embodiment, variable limiting: A rollerized cam follower 24, including a roller 28 rotatably mounted on a roller shaft 22 fixedly attached to follower 24, constructed in the form of a piston, slidable within release controller 36, is in rolling contact with valve cam 58 at intake cam face 62. Valve cam 58 typically contains an exhaust cam face 74. Cam follower 24 is hollow to accept a compression spring 28 and a portion of a fluid plug 25 and slidably engage valve lifter 50. Further, at one axial location along cam follower 24 is a radial conduit and an annulus 26, connecting the

inside of cam follower 24 with the inside of release controller 36. Controller 36 in the form of a hollow cylinder is slidably engaged with a limiter housing 52. Controller 36 has a controller annulus 38 connected to a return conduit 35 through a drain conduit 60 and a drain chamber 56. Controller 36 is pivotally connected to a controller drive link 32 by a link pin 34. A valve lifter 50 in the form of a stepped cylinder hollow at both ends has one end slidably engaged with both housing 52 and follower 24. Lifter 50 is hollow towards the follower end to accept spring 28 and a portion of fluid plug 25. Fluid plug 25 is connected by a radial conduit with a supply annulus 43 on lifter 50 and a supply conduit 44. Supply conduit 44 is connected to supply 38 through check valve 42 and supply pump 40. The hollow end of lifter 50, connected with an intake valve 48 through a spacer 46, is slidably engaged with housing 52. A valve spring 49 maintains closing force on valve 48. A step 51 between the outer diameters of lifter 50 abuts a step in limiter housing 52. A bypass conduit 41 connects with supply conduit 44. Conduit 41 returns hydraulic fluid through cutoff valve 45 and pressure relief valve 47 to supply 38. A limiting actuator, schematically represented by encircled letters LA, is to move link 32 and controlling the limiting and hence the speed. In the simplest case, it would represent a linkage system connecting link 32 to the accelerator pedal. In more sophisticated systems, it could represent electro-pneumatic or electro-hydraulic pistons, operated by the central control system described later.

Embodiment, differential shifting: Valve cam 58 is rotatably engaged between a main drive shaft 54 and a thrust bearing 64 and is in contact with a roller 28 at intake cam face 62. A bevel gear 68 meshes with a gear on valve cam 58 and a cam drive gear 66 fixedly attached to shaft 54. Gear 68 is rotatably mounted in a gear ring 72 on a bevel gear shaft 76. Ring 72 is rotatably mounted between drive gear 66 and bearing 64 and is pivotally pinned to a gear ring drive link 70. A shifting actuator is schematically represented by circled letters SA, to move link 70 and thereby shift between economy and power modes. Any number of known apparatus can be used to accomplish this, hydraulic or pneumatic pistons, shift levers, etc..

Operation, variable limiting: Pressurized hydraulic fluid is introduced through check valve 42 and conduit 44, completely filling the closed chamber that forms fluid plug 25 and interconnections thereto. Rotation of disk cam 58 drives cam follower 24, with periodic forces produced by the cam, through excursion path 78 of Fig. 5. Movement of follower 24 will be

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transmitted through the enclosed fluid to lifter 50, opening or closing valve 48. This movement will bring follower annulus 26 to overlap, or partially align with, controller annulus 38, completing a flowpath from plug 25 to return conduit 35. When this overlap occurs, the fluid in fluid plug 25 can escape and lifter 50 is free to drop. Once the overlap occurs it must be maintained until lifter 50 has returned to quiescent position. Spring 49 drives or biases valve 48 and lifter 50 to closed position. Follower 24 may still be moving towards the lifter but, will meet little resistance since fluid plug 25 is released.

Uncushioned descent of valve 48 would result in undesirable impact with the valve seat upon closure. The chamber formed between step 51 and the corresponding step on housing 52 will fill with hydraulic fluid as lifter 50 opens valve 48. As the valve closes, lifter 50 descends and fluid between the steps is forced through annulus 43 into fluid plug 25. When annulus 43 is closed off from the step chamber, a hydraulic cushion is formed. The diameters between step 51 and annulus 43 can be modified or shaped, limiting leakage to control the resistance of the cushion. The valve clearance, with lifter 50 and intake valve 48 in the closed position, is set by varying thicknesses of spacer 46.

Follower 24 reaches the extreme up position on curve 78 at about T3 in Fig. 5. Plug 25 is released and lifter 50 descends along curve 80. In prior art, curve 80 would also correspond to cam follower travel which is slaved to the cam drivetrain and would be built into the cam profile and, follower 24 would descend in the relatively short time period from T3 to T4. Supply pump 48 would need to be of sufficient size to refill fluid plug 25 during the T3 to T4 time period. The fluid pressure on follower 24 combined with the force from spring 28 must be sufficient to maintain follower 24 in contact with cam 58 during the descent. The pause in the extreme up position of follower 24, between T3 and T4 on curve 78 in Fig. 5, allows lifter 50 to descend to closed position. In the closed position supply annulus 43 in lifter 50 overlaps supply conduit 44 and fluid plug 25 can be refilled. If this were not the case, hydraulic fluid would flow continuously from the supply once the fluid had been released.

It is one feature of this invention that the descent of follower 24 has a prolonged duration such that it takes place in an extended time period from T4 to T5. Slowing the descent to roughly one fourth of the rate from T3 to T4. The extended descent of follower 24 requires that the descent of lifter 50 always occurs due to release of fluid plug 25 and not due to following the cam profile down, as in prior art. This means a smaller

pipng and pump 40 capacity than required by the prior art to maintain contact of follower 24 with cam 38.

The beginning of valve closure is determined when follower annulus 26 overlaps controller annulus 38. When annulus 38 is positioned the farthest from annulus 26, when annulus 26 is at quiescent or down position, it takes longer for them to move to overlap. Thus, valve 48 is open the longest duration and closure commences at time T3 in Fig. 2. Conversely, the shortest open duration occurs when annulus 38 is positioned closest to annulus 26 and closure commences at T1. The open time is determined by the relative quiescent positions of annulus 38 and annulus 26, which in turn is determined by the position of controller 36. Controller 36 can be positioned by moving drive link 32 with the limiting actuator LA. Valve closure can be selectively started for any intermediate time T2, from T1 to T3, producing lifter 58 descent along curve 82 in Fig. 5. Thus, the fuel air charge to the cylinder can be continuously and variably limited as it is by the throttle in a car. With a fuel saving difference, the throttling losses are eliminated.

Intake valve 48 can be deactivated to reduce active displacement or, to close the valves during stroke shift. For active valve operation, cutoff valve 45 remains closed and operation proceeds as described before. A signal from the engine control system opens valve 45. The signal could be an applied voltage if valve 45 is solenoid operated. Fluid plug 25 can now escape out conduit 41 through valve 45 and pressure relief valve 47, provided that the fluid pressure exceeds the relief valve setting. This pressure setting would have a minimum level to prevent excessive flow from supply pump 40 and a maximum level below the pressure needed to overcome valve spring 49 and open valve 48. Thus, when cutoff valve 45 is closed, the intake valve is active and when open the intake valve is deactivated.

The above system provides a reliable and relatively low cost hydro-mechanical limiting system, either for early or late intake valve closure. It needs only passive control for steady state operation. No timed signal is required for each valve cycle.

It is a further advantage of this invention, combining limiting with increased pre-ignition pressure, to improve combustion and eliminate or substantially reduce the higher hydrocarbon level. Limiting alone in prior art engines produces relatively higher hydrocarbon levels in the idle and lower partial load regions. A review of pre-ignition temperature T_{pi} and pressure P_{pi} profiles in Fig. 3 offers an alternative theory, to that expressed in the referred U.S. Pat. 4,765,288. As the load is reduced in a

throttled engine the pre-ignition temperature increases, whereas in a limited engine the temperature decreases. And, as the load is reduced, the pre-ignition pressure for both throttled and limited operation goes down, with limited going lower. At idle for limited operation, approximately 20 percent IMEP, where the maximum hydrocarbon production occurs, the absolute temperature is lower by 17 percent and the absolute pressure is lower by 25 percent. Either of these relative conditions can have a negative effect on the quality of combustion and hence contribute to higher hydrocarbon production.

In this invention, the pre-ignition pressure at idle and in the lower partial load regions is approximately twice that of either throttled or limited engines. And, the pre-ignition temperature at idle has been almost fully restored to throttled levels. Both changes are towards decreased hydrocarbon production and combined may reduce it below throttled levels.

A further advantage is to reduce and possibly eliminate the CO and HC emissions that comes from the excessively rich mixture produced by high manifold vacuum during deceleration and idle. This vacuum rapidly evaporates fuel condensed on the manifold walls. In a limited engine, there is no manifold vacuum. The manifold pressure is essentially constant at atmospheric pressure. No vacuum, no rich mixture.

Another advantage is to reduce NO_x emissions by reducing their production during combustion. The residual gas, left in the cylinder from the previous cycle, acts as a diluent in the new unburned mixture. The absolute temperature reached after combustion varies inversely with the burned gas mass fraction. It is known that increasing this burned gas fraction reduces NO_x emission levels substantially. In economy mode, where most engine operation will occur, and possibly all in an economy mode only engine, the exhaust clearance volume is larger than the combustion clearance volume. The larger volume leaves more unburned gas in the cylinder and would have the effect of decreasing the NO_x emissions.

Operation, differential shifting: To phase shift the two strokes, or the required 180 degrees, the relationship of disk cam 58 to main drive shaft 54 must shift 180 degrees on a four stroke cam. If desired, the valves are then deactivated as previously described. Prior to the phase shift, gear ring 72 is stationary. Drive gear 66 rotates with drive shaft 54 and meshes with the bevel gear 68. Gear 68 meshes with disk cam 58, driving it in the opposite direction. The pitch diameters of the gear on cam 58 and drive gear 66 are equal. Therefore, as shift actuator 5A moves link 70, driving gear ring 72 circumferentially through 90 degrees, the

relationship between disk cam 58 and drive shaft 54 is shifted the required 180 degrees. The valves are reactivated and the phase shift is complete. The exhaust valves in prior art engines have their cam profiles on the same disk cam but in a different location. This is the case here and as intake valve cam face 62 is shifted, exhaust valve cam face 74 is also shifted. The same exhaust profile can be used since exhaust need not be limited, although changeable for timing variation. Another object can be achieved by modulating the two quiescent positions of gear ring 72 with shift actuator 5A. Specifically, the timing for both the intake and exhaust valves can be advanced or retarded the same amount together.

Description/Operation, incremental embodiment, Fig. 8 & 9

Embodiment, incremental limiting: A rollerized cam follower 24A is constructed in the form of a piston on the end of a smaller shaft, slidable within a limiter housing 52A. Follower 24A is maintained in contact with a valve cam 58A, by a compression spring 28A and pressure from hydraulic fluid in a chamber 120. A release adjuster 118, constructed in the form of a piston, is adjustably affixed to follower 24A. The other end of spring 28A is in contact with housing 52A. A valve lifter 58A is constructed in the form of a piston on the end of a smaller shaft, slidable within housing 52A and a cushion adjuster 100, partially exposed to fluid in chamber 120 and, maintained by fluid pressure in contact with valve 48. Valve 48 is springably loaded towards the closed position by valve spring 49. A cushion chamber 101 is formed between lifter 58A and adjuster 100. Hydraulic fluid is supplied as in Fig. 4, through supply conduit 44A. Cushion adjuster 100 has an internal cushion annulus 106 connected through a conduit to chamber 120 and is adjustably affixed to housing 52A. Release adjuster 118 has a release face 108 on the chamber 120 end. Limiter housing 52A has two or more annuli on the interface with release adjuster 118, a power annulus 102 connected to a return conduit 35A and, an intermediate annulus 104 connected through a release valve 122 to drain. The axis of lifter 58 in this embodiment is shown behind spring 28A and follower 24A and, all are exposed to the fluid in chamber 120.

Embodiment, trip shifting: Valve cam 58A is rotatably engaged between a main drive shaft 54A and a bearing 64A and is contacted by rollerized cam follower 24A. A trip lever 112 is rotatably mounted on a shaft 116 which is fixedly attached to cam 58A and has two positions of engagement, P1 and P2, with a trip key 114, a stop ring 110 and a detent pin 124. Stop ring 110 is an assembly of an inner ring and an outer ring fixedly attached

together through a shock absorbing material, such as molded rubber. Trip key 114 has two positions of slidable engagement in a stationary housing, K1 and K2. Trip lever 112 has two positions determined by detent pin 124 which is held into a detent in lever 112 by the force of detent spring 126. Trip lever 112 also has a tab 128 that projects into the position of trip key 114 during rotation if, trip key 114 is in position K2. Stop ring 118 is fixedly attached to shaft 54A and engages stop face 138 on lever 112 so as to drive cam 58a. A shifting actuator is schematically represented by the encircled letters SA-A, to move trip key 112 for shifting between economy and power modes. A number of known apparatus can perform this function, like the actuator in Fig. 4. Fig. 9 is a sectional view of the trip lever, clarifying the two positions.

Operation, incremental limiting: Functional operation is the same as Fig. 4 except lifter 58A and follower 24A axes are not coincident and the limiting is not continuously variable, occurring only at fixed positions. As follower 24A is driven through the excursion path in Fig. 2, fluid is displaced in closed chamber 120. The incompressible displaced fluid raises lifter 58A accordingly. Lifter 58A motion continues until release face 188 exposes or overlaps the intermediate annulus 184 to chamber 120. If valve 122 is open the fluid is released and lifter 58A is driven down by the force of valve spring 49, closing valve 48. If release valve 122 is closed, nothing changes and follower 24A continues until face 188 exposes or overlaps the power annulus 182. Annulus 182 is always connected to return conduit 35A, releasing the fluid to close valve 48, so that the protracted refill may be used. Depending on the distance required between annulus 182 and annulus 184, opposing segments of annuli could be used to stagger them closely. Release valve 122 is passive except when changing operating modes. Either open for economy or closed for power mode. A hydraulic cushion is formed in chamber 181 when the shaft of lifter 58A penetrates adjuster 180 far enough to close off annulus 186. Variations in manufacturing tolerances or strength of cushion can be compensated for by moving adjuster 180 relative to housing 52A. Release adjuster 118 can also be adjusted relative to follower 24A to compensate for manufacturing tolerances assuring valve 48 closure at the proper time.

The addition of another annulus and valve, similar to annulus 184 and valve 122, offers other levels of limiting. A similar annulus and valve, located overlapping the quiescent position of the large face on follower 24A, could be used to retard the opening of a valve until the overlap is closed. And another annulus, connected to a return conduit, located to

just overlap the maximum desired open position of the large face of lifter 58A, could be used to limit the maximum opening of valve 48.

Since continuously variable speed control is required, a throttling system would be used as in the prior art. During power operation, the ITE would follow the throttled profile in Fig. 3. During economy operation, the ITE would peak at the same point as the invention profile but, throttle down from there along the dashed line shown.

Operation, trip shifting: Prior to the shift, stop ring 110 is engaged with trip lever 112, shown in position P1, driving valve cam 58A with main drive shaft 54A. As lever 112 is moved past the stationary trip key 114 in position K1 shown, no interaction occurs. Detent pin 124, forced into the detent in lever 112 by detent spring 126, holds lever 112 in position. Shifting 180 degrees, between economy and power position, is accomplished by shift actuator 8A-A moving key 114 to position K2, shown in dashed lines. As lever 112 rotates past key 114, key 114 will now strike tab 128, rotating lever 112 on shaft 116 to the other detent position P2, shown in dashed lines in Fig. 9. This momentarily disconnects cam 58A from shaft 54A. Undriven cam 58A will slow until stop face 130 on lever 112 engages the opposite stop on ring 110. The shock absorbing material in stop ring 110 will absorb the impact. Cam 58A will continue to rotate with shaft 54A except, the relative positions have changed 180 degrees, phase shifting between economy and power modes. The phase shift is complete. Returning the position of trip key 114 to K1 would cause it to stroke the leading edge of trip lever 112, rotating it to position P1. Stop ring 110 would re-engage trip lever 112 restoring the former mode.

Description/Operation, Flaplatch, Fig. 6 & 7

Embodiment, flaplatch, fig. 6: A conventional rocker arm 14: is driven by a push rod 94 and drives valve 48; rocks about fulcrum 90; is maintained in contact with valve 48 and pushrod 94 by the force transmitted thru fulcrum 90 from fulcrum spring 92 which pushes against a supporting housing; and is provided with locational stability by a shaped surface on fulcrum 90 held in contact by that force. Fulcrum 90: is pinned to a supporting housing and may rotate about the pin, but is otherwise confined; and has abutting face 96 and displacing surface 88, both for engagement with flaplatch 16. A flaplatch 16: is pivotally and adjustably supported by a supporting housing and is restrained from separating therefrom; has a first position as shown engaging abutting face 96 and displacing surface 88; and has a second position, determined when displaced out of engagement

by surface 88, that is in intimate proximity with the action end of electromagnetic actuator 15; and can be held in the second position by the magnetic attraction of electromagnetic actuator 15. An electromagnetic actuator 15i has a permanent magnet 15a and a coil 15b that when electrically energized counters and thus reduces the magnet attraction holding flaplatch 16; and is mounted on a supporting housing. A flap spring 86 mounted on housing 52B biases flaplatch 16 towards engagement with fulcrum 90 with insufficient force to overcome the holding force of a non-energized electromagnetic actuator 15, and sufficient force when it is energized. A valve cam 58B drives pushrod 94 thru follower 24B. The excursion of cam follower 24B in response to rotation of cam 58B is shown in Fig. 7 and has a base line 169, an economy lobe 168 above and dip 167 below base line 169. A second dip and lobe, corresponding to operation two strokes later, are shown and will be explained later.

Operation, flaplatch, fig. 6: The rocker 14 acts in the manner of a conventional rocker when flaplatch 16 is pressed by spring 86 into engagement with fulcrum 90. The periodic forces pressing rocker 14 onto fulcrum 90 are reacted thru the pivot pin, fulcrum spring 92, and thru abutting surface 96 into the end face of flaplatch 16. And, thru flaplatch 16 into the supporting housing. This is a valve enabling position for flaplatch 16 in which fulcrum 90 is fixedly supported and the normal positive motion of the pushrod 94 translates into valve operation.

Disabling valve operation occurs as follows: When valve cam 58B rotates to where cam follower 24B rolls into dip 167, the follower's descent produces a "negative" motion for push rod 94. Both pushrod 94 and follower 24B are being driven negatively by the reactive force from fulcrum spring 90. This negative action results in the fulcrum 90 pivoting in the direction away from flaplatch 16, but not so far as to no longer overlap flaplatch 16. This action brings displacing surface 88 to bear on the corresponding surface of flaplatch 16, displacing flaplatch 16 into a position in close proximity of electromagnetic actuator 15. This is the second position of flaplatch 16 and is potentially a holding position depending on the energized state of electromagnetic actuator 15. If the valve is to be disabled, then the coil 15b will not be energized, and the flaplatch 16 will remain held by the force of permanent magnetic 15a. In this holding position, abutting face 96 is misaligned with the corresponding end face on flaplatch 16. This misalignment permits abutting face 96 on fulcrum 90 to swing past the end face on flaplatch 16 when positive action is produced by the lobe on valve cam 58B. Fulcrum 90,

yields resiliently to this positive action, disabling the normal valve operation so long as flaplatch 16 is held.

Enabling valve operation from the potential holding position requires energizing coil 15b into the enabling state and releasing the hold on flaplatch 16. Releasing the hold at least for the short period when fulcrum 98, returning to quiescent position from the extreme negative action position where the cam follower is at the bottom of dip 58a, allows flaplatch 16 as driven by flap spring 86 to slide down displacing face 88 into alignment and engagement with abutting surface 96. This is the window for re-activating valve 48.

It is one feature of this invention that flaplatch 16 is displaced or positively driven out of engagement each cycle. Prior to each lobe that may be selected. Displaced by means other than the holding or capturing device. Actuator 15 in this case. This makes for high speed operation, since the holding device need only be on or off as the flap is "presented" and does not have to move anything, or overcome it's inertia. As would be the case in a solenoid driven latch, for example.

Fig. 7 graphs the excursion path of cam follower 24B and pushrod 94 with respect to the main shaft or cam 58B rotation. Cam 58B has two lobes that produce the excursion path shown: a power lobe 166 for normally aspirated power mode; and a charge limiting economy lobe 168.

The purpose of the second dip and lobe on valve cam 58B is apparent when considering that one lobe can be turned on and the other off, or visa versa to accomplish a two stroke shift. At the very least, valve operation can be changed by shifting to another lobe and combusting at the next clearance volume. Even in constant clearance volume crank engines.

The ability to electrically switch on or off each valve opening, coupled with a selection of opening profiles, allows engine operation to be orchestrated by a computer chip. In the context of this cam engine, with two appropriately placed lobes, shifting from one to the other effects a valve shift from economy mode to power mode. And, essentially only by modifying the chip, the selection includes: the fuel saving rolling deactivation of cylinders called "skipfire"; deceleration cutout of cylinders; engine braking by cutting our intake operation and activating both exhaust lobes. Or, a two stroke phase shift from high to low speed cam profiles or to accomodate Miller supercharging; etc..

Valve 48B and push rod 94 action in this case is shown parallel to the main shaft 54B. However, follower action could also be radially disposed against a rocker with a right angle bend, and the resilient support at 45

degrees. Another form of this invention, having an overhead cam driving the center of the rocker, would have displacing surface 88 and abutting face 96 an integral part of the end of the rocker that is opposite the valve. Flap latch 16 then engages rocker 14 directly and becomes the functional fulcrum, and may even pivot slightly during valve action. The equivalent of fulcrum 98 would engage the rocker near the latching end, retaining only the spring loading and locating functions. In this and other forms, fulcrum 98 need not pivot and could be disposed to slide linearly, etc.. In another form the function of displacing the latch could be done by another linkage and/or cam.

There are numerous other forms possible within the scope of this invention. Electromagnetic actuator 15 can be revised in order to hold when energized. A spring driven ratchet pawl, or hook, could capture and hold, or not capture and enable the flap. A major determinant is the speed, or the rate of cycling required, which is enhanced by displacing the flap each cycle. Slower acting forms, such as for cylinder cutout with a single dip and lobe, can omit the displacing action, displacing surface 88, and dip 58a, and, simply be arranged to pull the flap out for disabling, or push it back for enabling. The actuators could be biased or double acting, solenoid, hydraulic, or vacuum actuators, etc..

Supercharging

A synergistic effect occurs when limiting is used to control the output of a supercharged engine. Controlling the charge by limiting directly controls the operating compression ratio. In Fig. 2, at C1 the pressure and temperature conditions when the valve closes on the intake stroke are nominally restored at the same piston position on the compression stroke, at C2. The volume at C2 equals the volume at C1 and, is essentially unthrottled or at atmospheric pressure. As limiting varies this volume the operating compression ratio is proportionately varied.

Any reduction in compression ratio caused by supercharging is not necessary when limiting is used, if two more elements are added. First, the supercharging pressure must be sensed, or computed based on known engine characteristics. Second, the maximum limiter position reduced, or limited back, in accordance with the supercharger pressure. Otherwise, stepping on the accelerator would result in pre-ignition firing. The combined compression ratio would always equals the combination of the full supercharging compression ratio and an appropriately reduced operating compression ratio. The supercharger compression is always fully utilized

and the cylinder compression adjusted. The result, as the engine accelerates into the supercharged speeds, is that the blowdown work recovered by the supercharger is now fed back into the engine. This work replaces compression work previously done by the piston, and thus adds directly to shaft output. This increases the output per unit displacement and decreases SFC by more than prior art supercharging and limiting separately, the synergistic effect. According to Zinner in Supercharging of IC Engines the increase in output can be from 25 to 40 percent. The higher compression from supercharging can replace cylinder compression.

It should be noted, that these feedback advantages apply to all displacement type, or expansible chamber, engines that can have their maximum charges throttled back or limited back: internal combustion, external combustion, other forms of heating, compression ignition or spark ignition. The substantial work used for compression in a diesel engine could be partially replaced by work recovered from the exhaust.

Description/Operation, Control System, Fig. 10

A system to control the continuously variable limiting arrangement in Fig. 4, is shown in Fig. 10. It is shown schematically and illustrates the controls relevant to this invention. In prior art and for this invention, this system would probably contain an electronic control unit or ECU in a control system 164, coupled with an array of mechanical, electrical, pneumatic and hydraulic devices for sending, receiving and actuating. The ECU would receive input signals from respective sensors, representative of engine speed, shaft position, or RPM, loading demand on the engine, for example derived from a potentiometer coupled to an accelerator pedal 154, supercharger speed and pressure, oil and water temperatures, etc.. The ECU would contain stored data, representative of engine operating characteristics relative to various variable input parameters, and provide appropriate output signals, such as selecting the appropriate clearance volume. And, changing the ignition or combustion timing in accordance with the selected clearance volume. These signals would control the shift actuator through a line 158, to put the engine in economy or power mode and deactivate the valves during shifting by opening the cutoff valve through a line 162. Depending on mode, the accelerator stop 156 would be positioned to avoid overcharging the cylinders. The speed control actuator would control engine speed through a line 160 by controlling the open duration of the intake valves.

The system to control the dual lobe arrangement in Fig. 6 would be the

same as for Fig. 4 except: the speed control actuator would operate a throttle as in the prior art; the accelerator stop would be omitted since the limiting function is built into the two different lobe shapes on the valve cam; and an appropriately timed signal would be sent during each valve each cycle to select the active lobe.

The system to control the arrangement in Fig. 8 would be the same as for Fig. 6 except accelerator stop 156 would be eliminated and the function of stop 156 accomplished by release valve 122, shown in dashed lines. Additional release valve positions, with different levels of limiting, could be optionally provided for supercharged compression compensation or work feedback, etc..

Summary

Increasing the operative options in an engine is the ultimate advantage of this invention. Having more modes offers a selection of capabilities heretofore prohibited in a single engine. Either as multiple modes enhancing performance for a single fuel, or single dedicated modes enhancing the performance for multiple fuels. Phase shifting the operating stroke accesses these modes of operation. Variations in a piston drive cam offers a selection of compression and expansion ratios. Variable valve control offers a selection of both intake or exhaust displacements. Either continuous or incremental valve control enables selecting the degree of throttling losses. One option being essentially none. Limiting back in a higher compression ratio mode enables maintaining the knock-limited compression ratio at a reduced intake displacement, yielding a higher indicated thermal efficiency at part power. Limiting back in a supercharged engine enables the feedback of supercharger compression into the engine. Many of these options are applicable to any single or dual fuel expansible chamber engine, broadly defined as one that expands a chamber with compressed fluid to produce a useable output.

The ability to phase shift during operation enables the achievement of several long sought goals: The maximum pre-ignition pressure ratio R_{pi} of 18 is maintained at maximum economy power, instead of dropping to 13 and lowering efficiency as in a throttled engine. Avoiding throttle associated losses at approximately 59 percent power. Operating on a greater expansion ratio cycle at part load, makes the efficiency at part load better than at full load, reversing the prior art relationship. Reducing displacement without shutting off cylinders maintains operating temperature and thus performance. Reducing variations in manifold vacuum that produces the rich

mixture during deceleration and idle; increasing pre-ignition pressure and temperature at idle and lower part load; and increasing burned gas mass fraction in economy mode; all point at reducing HC, CO and NOx emissions.

Another advantage was apparent from a test performed by the writer. A vacuum gauge was connected to the intake manifold of the 387 Oldsmobile engine. The car was driven through various city and suburban conditions using normal speed, acceleration and deceleration. The vacuum varied from 18 to 28 inches of mercury. The engine operated at all times at a power level that would fall within economy mode. The proverbial car driven by an old lady schoolteacher would never be shifted into power mode. In a practical sense, the power mode could be treated as a passing gear, with the bulk or even all of the operation occurring in the more fuel efficient and less emissive economy mode.

Each of the three valve arrangements has it's own merits: Fig. 4, with continuously variable limiting and modulation of timing, offers sophisticated yet passive control where throttling and the associated losses can essentially be eliminated. Fig. 8 with speed control by throttling and incremental limiting plus the phase shift, is also passive, and further offers: optional increments of limiting for idling, Miller supercharging, etc.; shorter engine length; an easier to manufacture radial cam profile; improved adjustability; and adaptability to splayed valves radially oriented in a spherically radiused head, reducing the critical surface to volume ratio. Fig. 6 has two modes of valve operation built into each two lobe cam profile. Shifting of the cam to shaft relationship is not required. Instead, the desired lobe is activated and the other deactivated, or visa versa.

The high speed electrical valve control of the flaplatch system enables practical valve and cylinder orchestration during an engine cycle. This enables skipfire, deceleration cut-out, engine braking, etc..

Limiting with supercharging enables recovering exhaust energy as cylinder compression. This makes Miller supercharging of an automotive engine practical and can add 25 to 40 percent to output.

Many modifications and variations of the disclosed features of this invention are possible. For example: the variable release design of Fig. 4 can be adapted to the non-coincident follower 24A and lifter 58A axes design of Fig. 8, making for a shorter engine or for better adjustability, etc.; any of the three valve arrangements can be incorporated into spark or compression ignition engines of conventional in-line, V, or other designs to provide variable valve control, including those with constant

clearance volumes; the dual lobe shift could be effected by valvably releasing a fluid plug. The double ended pistons of Fig. 1 could be single ended. An economy mode only engine is possible. A multi-lobe valve can offers multiple profiles, but needs a phase shift to access them. A single-lobe-cam shaft shift in a cam engine could match compression ratios with fuels in a dual fuel engine. It is to be understood, therefore, that the invention can be practiced otherwise than as specifically described.

The bottom line for any invention, what it can achieve, is best stated for this invention in an automotive context. Using the road test results of the referred U. S. Pat. 4,288,451, where a 23 percent increase in MPG was measured, against a calculated increase in indicated thermal efficiency for the tested engine, and, calculating the increase in the same efficiency using the same method for the invention engine, excluding supercharging and regeneration, the projected increase in MPG is

---- 56 percent, ----

without reducing the maximum power of the engine.

It will be apparent to anyone familiar with the prior art, that this is not just an improvement of existing art, but a fundamental change in the way an engine is operated. It is a pioneering invention representing a breakthrough in engine technology, in one of the most competitive and crowded fields. As such it deserves the broadest interpretation of the following claims as to the heart and the essence of this invention.

Claims: I claim:

1. In an expansible chamber engine having at least one cylinder, said cylinder having a piston, said piston defining in part a clearance volume at top dead center, said clearance volume being one of a continuous series of clearance volumes, the position of said piston at top dead center being effected by a piston drive cam, an improvement comprising: means for continuously alternating said clearance volume between a maximum and a minimum clearance volume, said minimum clearance volume being sufficient to contain a power producing charge.

* 2. The engine of claim 1, further including in an engine wherein said cylinder has a functional cycle following the piston top dead center position, an improvement comprising: means for shifting said functional cycle to follow a subsequent clearance volume.

* * 3. The engine of claim 2, wherein said means for shifting in an engine having at least one valve controlling fluid flow to or from said chamber, drivetrain means for effecting operation of said valve in response to periodic forces from a valve cam, and driving means driving said valve cam, comprises: means for changing the rotational relationship of said valve cam to said driving means to effect the shift for said valve.

* * * 4. The engine of claim 3, further including in an engine having means for providing a fluid plug hydraulically coupling said drivetrain means, means for supplying pressurized hydraulic fluid to said fluid plug, and means for draining hydraulic fluid, said valve being biased towards closure, an improvement comprising:

a plug port connected to said fluid plug;

a release port connected to said means for draining hydraulic fluid; and
means for closing said valve effected by the advent of an overlap

connecting said plug port and said release port, said overlap releasing fluid from the hydraulic coupling thru the connection and thus permitting the valve to close, said advent effected by relative motion between said plug port and said release port, said relative motion effected by said valve cam.

* * * * 5. The engine of claim 4, further including a valvable connection interposed between the drain and a port, said port selected from the group consisting of said release port and a second release port, and means for

selectively maintaining and releasing said hydraulic coupling upon said overlap in accordance with the state of said valvable connection.

* * * 6. The engine of claim 4, further including a cam follower following said valve cam, and means for returning said cam follower to quiescent position after the latest effective closing point of said valve.

* * * 7. The engine of claim 4, further including means for varying the position of said release port.

* * 8. The engine of claim 2, further including in an engine having at least one valve controlling fluid flow to or from said chamber, an improvement comprising:

a valve cam having a plurality of lobes, the timing of said lobes corresponding to the timing of said series of clearance volumes; and means for selectively enabling and disabling the valve operation effected by each of said lobes.

* * * 9. The engine of claim 8, wherein said means for selectively disabling and enabling includes means for changing the fulcrum of a rocker arm in the drivetrain, comprising:

support means adapted for attachment to said engine;

locating means mounted on said support means, said locating means defining a pivot surface adapted to contact the rocker, the mounting of said locating means adapted to define the motion of said pivot surface, said locating means further being biased to contact the rocker with sufficient force to overcome clashing in the valve drivetrain;

abutting means adapted to transmit lateral support to said rocker, said lateral support effecting a fulcrum to enable normal opening of said valve, said abutting means defining a first abutting surface;

latch means defining a second abutting surface for contacting said first abutting surface and adapted to react the drive train forces which effect normal opening of said valve, said latch means being mounted on said support means and adapted to enable said second abutting surface to move into and out of alignment with said first abutting surface, the movement of said second abutting surface being essentially towards and away from both said first abutting surface and the axis of the reactive force thru said pivot surface on said rocker;

actuation means selectively operative to switch between a disabling state

and an enabling state, said disabling state providing for holding said latch means in a disabling position misaligning said abutting surfaces and effecting disablement of said valve by allowing said abutting surfaces to move past each other in response to said drivetrain forces, said enabling state providing for driving said latch means to an enabling position aligning said abutting surfaces and effecting normal valve opening and closing in response to said drivetrain forces, said actuation means further including unlatching means providing for moving said latch means from said enabling position to said disabling position whereupon being selectively held or driven to said enabling position, said moving effected each engine cycle prior to the advent of each of said lobes.

* * * * 10. The device of claim 9, further including in said unlatching means a dip in said valve cam, said dip being below the base line of the cam profile prior to each lobe for providing negative rocker motion to effect said moving.

* * * * 11. The device of claim 9, wherein the means for holding said latch means in a disabling position comprises an electromagnetic actuator.

12. An improved valve cam for an expansible chamber engine having a valve controlling fluid flow to or from said chamber, a cam follower following a valve cam, drivetrain means effecting opening of said valve in response to said valve cam, said drivetrain means including means for closing said valve before the return of said cam follower to quiescent position, comprising: means for returning said cam follower to quiescent position after the latest effective closing point of said valve.

13. A method in an expansible chamber engine having at least one cylinder, said cylinder having a piston, said piston defining in part a clearance volume at top dead center, said clearance volume being one of a continuous series of clearance volumes, said cylinder having a functional cycle following the piston top dead center position, comprising the step of: shifting said functional cycle to follow a subsequent clearance volume.

* 14. The method of claim 13, wherein said shifting step includes shifting to a clearance volume having a different volume.

* * 15. The method of claim 14, wherein said shifting step in an engine having at least one valve controlling fluid flow to or from said chamber, a valve cam having a plurality of lobes, the timing of said lobes corresponding to the timing of said series of said clearance volumes, and means for selectively enabling and disabling valve the operation effected by each of said lobes, includes the steps of:

disabling valve operation for the lobe corresponding to said top dead center position; and

enabling valve operation for the lobe corresponding to said subsequent clearance volume.

* * 16. The method of claim 14, wherein said shifting step in an engine having at least one valve controlling fluid flow to or from said chamber, drivetrain means for effecting operation of said valve in response to periodic forces from a valve cam, and driving means driving said valve cam, includes the step of: changing the rotational relationship of said valve cam to said driving means to effect the shift for said valve.

* * * 17. The method of claim 16, further including in an engine having means for providing a fluid plug hydraulically coupling said drivetrain means, means for biasing said valve towards closure, means for supplying pressurized hydraulic fluid to said fluid plug, means for draining hydraulic fluid, a plug port connected to said fluid plug, a release port connected to said means for draining hydraulic fluid, and means for maintaining said hydraulic coupling when said plug and release ports are disconnected, the step of: closing said valve by the advent of an overlap connecting said plug port and said release port, said advent effected by relative motion between said plug port and said release port, said relative motion effected by said valve cam.

* * * * 18. The method of claim 17, further including the step of selectively maintaining and releasing said hydraulic coupling upon said overlap in accordance with the state of a valvable connection, said valvable connection interposed between the drain and a port, said port selected from the group consisting of said release port and a second release port, .

* * * * 19. The method of claim 17, further including the step of varying the position of said release port.

* 20. The method of claim 13, wherein said shifting step in an engine having at least one valve controlling fluid flow to or from said chamber, a valve cam having a plurality of lobes, the timing of said lobes corresponding to the timing of said series of clearance volumes, and means for selectively enabling and disabling the valve operation effected by each of said lobes, includes the steps of:

disabling valve operation for the lobe corresponding to said top dead center position; and
enabling valve operation for the lobe corresponding to said subsequent clearance volume.

* 21. The method of claim 20, wherein said enabling and disabling steps in said means for selectively enabling and disabling further includes the steps of:

moving a latching means from an enabling position to a disabling position, said latching means in the enabling position effecting support against a fulcrum surface of a rocker arm, said support enabling operation of said valve in response to the valve cam lobes, said latching means in the disabling position having effected the removal of said support, said removal permitting yielding at said fulcrum surface, said yielding sufficient to disable operation of said valve in response to said valve cam lobes, the movement of the engaging end of said latching means being essentially away from both the engagement and the axis of the reactive force thru said fulcrum surface, said movement occurring prior to the advent of each of said lobes to enable selection;
holding said latch means in said disabling position until enablement is selected; and
returning said latch means to said enabling position upon selection of enablement.

* * 22. The method of claim 21, wherein said moving step includes the step of a cam follower descending into a dip in said valve cam, said dip being below the base line of the cam profile prior to each of said lobes, the descent providing a negative rocker motion effecting the movement of said latch means thru said moving step.

* * * 23. The device of claim 21, wherein said holding step includes the

step of holding said latch means in a disabling position with an electromagnetic actuator.

24. A device for changing the fulcrum of an engine valve rocker arm to selectively disable and enable the valve, said device comprising:
support means adapted for attachment to said engine;
locating means mounted on said support means, said locating means defining a pivot surface adapted to contact the rocker, the mounting of said locating means adapted to define the motion of said pivot surface, said locating means further being biased to contact the rocker with sufficient force to overcome clashing in the valve drivetrain;
abutting means adapted to transmit lateral support to said rocker, said lateral support effecting a fulcrum to enable normal opening of said valve, said abutting means defining a first abutting surface;
latch means defining a second abutting surface for contacting said first abutting surface and adapted to react the drive train forces which effect normal opening of said valve, said latch means being mounted on said support means and adapted to enable said second abutting surface to move into and out of alignment with said first abutting surface, the movement of said second abutting surface being essentially towards and away from both said first abutting surface and the axis of the reactive force thru said pivot surface on said rocker;
actuation means selectively operative to switch between a disabling state and an enabling state, said disabling state providing for holding said latch means in a disabling position misaligning said abutting surfaces and effecting disablement of said valve by allowing said abutting surfaces to move past each other in response to said drivetrain forces, said enabling state providing for driving said latch means to an enabling position aligning said abutting surfaces and effecting normal valve opening and closing in response to said drivetrain forces, said actuation means further including unlatching means providing for moving said latch means from said enabling position to said disabling position.

* 25. The device of claim 24, wherein said unlatching means effects said moving each engine cycle prior to the advent of at least one lobe on a valve cam, said valve cam being the driver of said valve drivetrain.

* * 26. The device of claim 25, further including in said unlatching means

a dip in said valve cam, said dip being below the base line of the cam profile prior to said lobe for providing negative rocker action to effect said moving.

* * * 27. The device of claim 25, wherein the means for holding said latch means in a disabling position comprises an electromagnetic actuator.

28. In a valve control system for an expansible chamber engine having at least one valve controlling fluid flow to or from said chamber, said valve being biased towards closure, a valve cam, drivetrain means for effecting operation of said valve in response to periodic forces from said valve cam, means for providing a fluid plug hydraulically coupling said drivetrain means, and means for supplying pressurized hydraulic fluid to said fluid plug, and means for draining hydraulic fluid, an improvement comprising:
a plug port connected to said fluid plug;
a release port connected to said means for draining hydraulic fluid; and
means for closing said valve effected by the advent of an overlap connecting said plug port and said release port, said overlap releasing fluid from the hydraulic coupling thru the connection and thus permitting the valve to close, said advent effected by relative motion between said plug port and said release port, said relative motion effected by said valve cam.

* 29. The system of claim 28, further including a valvable connection interposed between the drain and a port, said port selected from the group consisting of said release port and a second release port, and means for selectively maintaining and releasing said hydraulic coupling upon said overlap in accordance with the state of said valvable connection.

* 30. The system of claim 28, further including a cam follower following said valve cam, means for returning said cam follower to quiescent position after the latest effective closing point of said valve.

* 31. The system of claim 28, further including means for varying the position of said release port.

SHEET 1 OF 6

FIG. 1

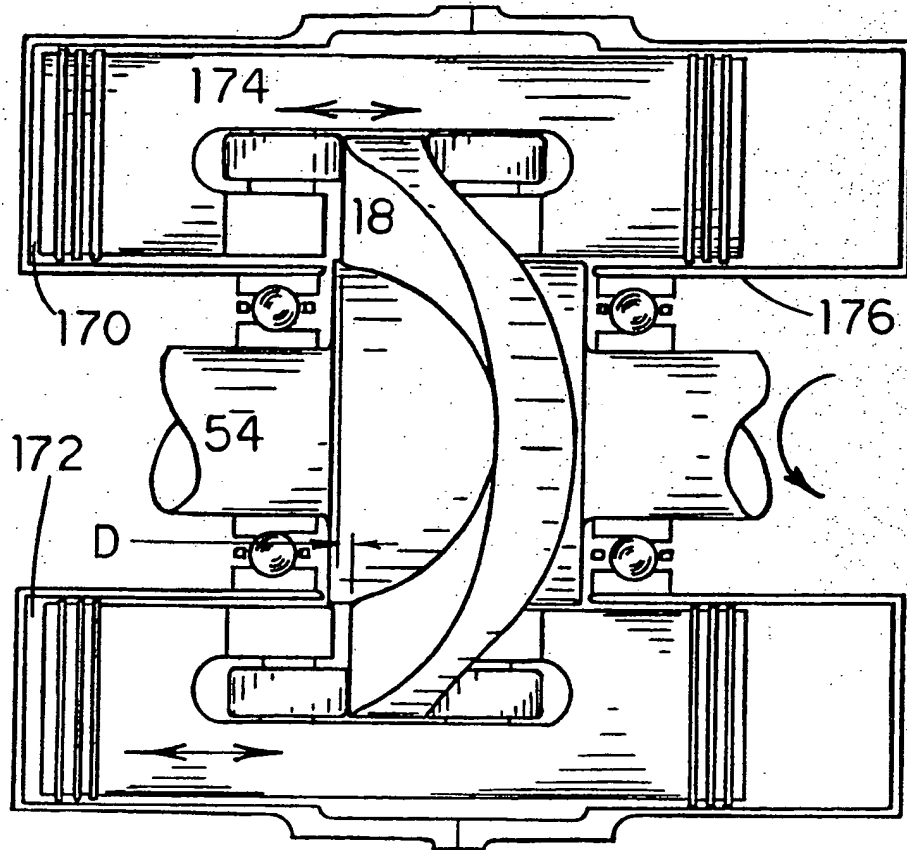
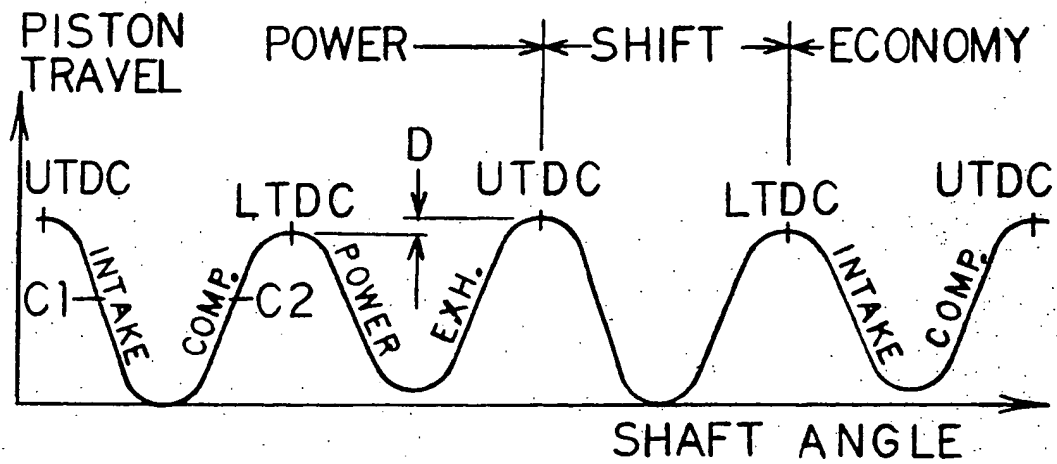


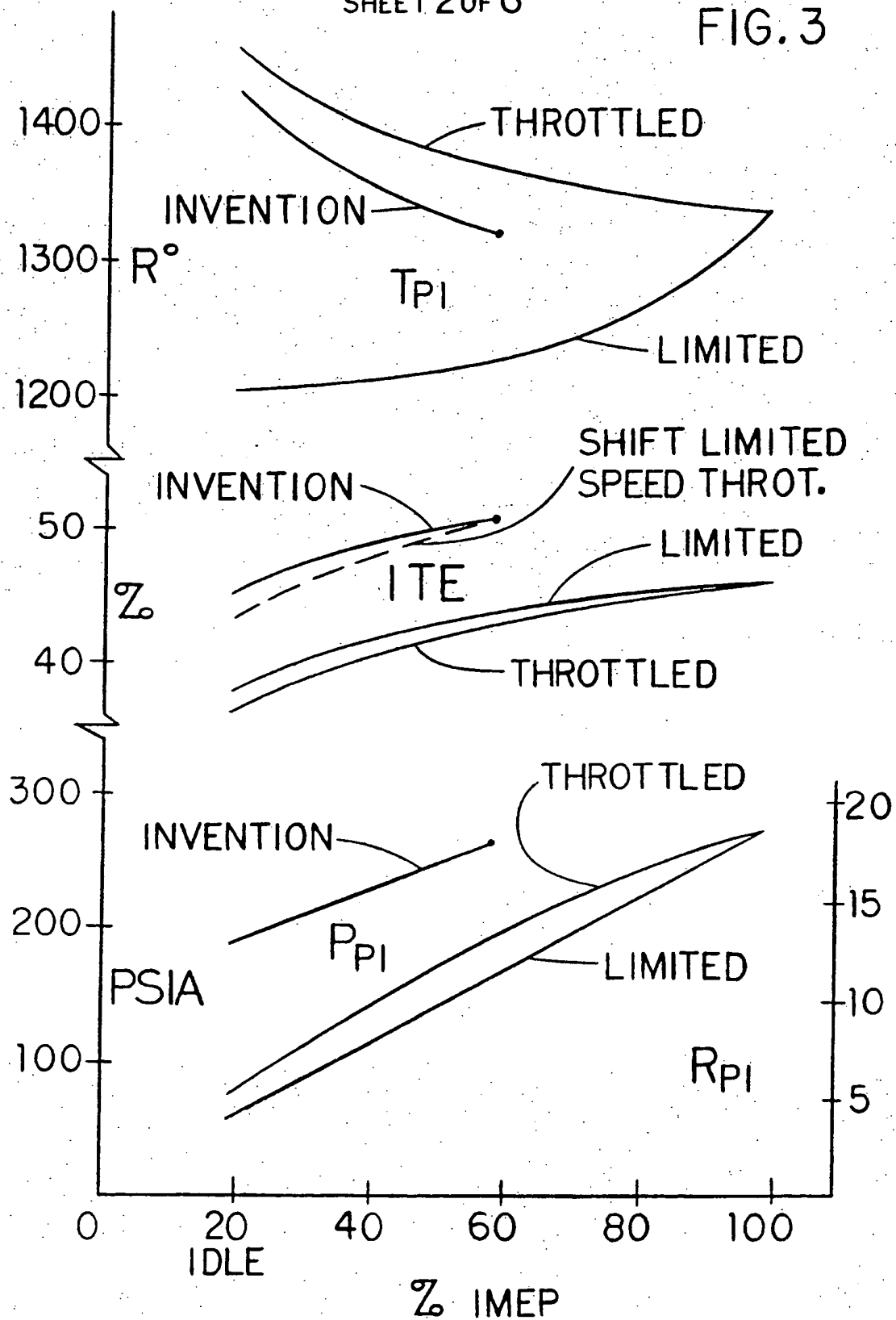
FIG. 2



SUBSTITUTE SHEET

SHEET 2 OF 6

FIG. 3



SUBSTITUTE SHEET

SHEET 3 OF 6

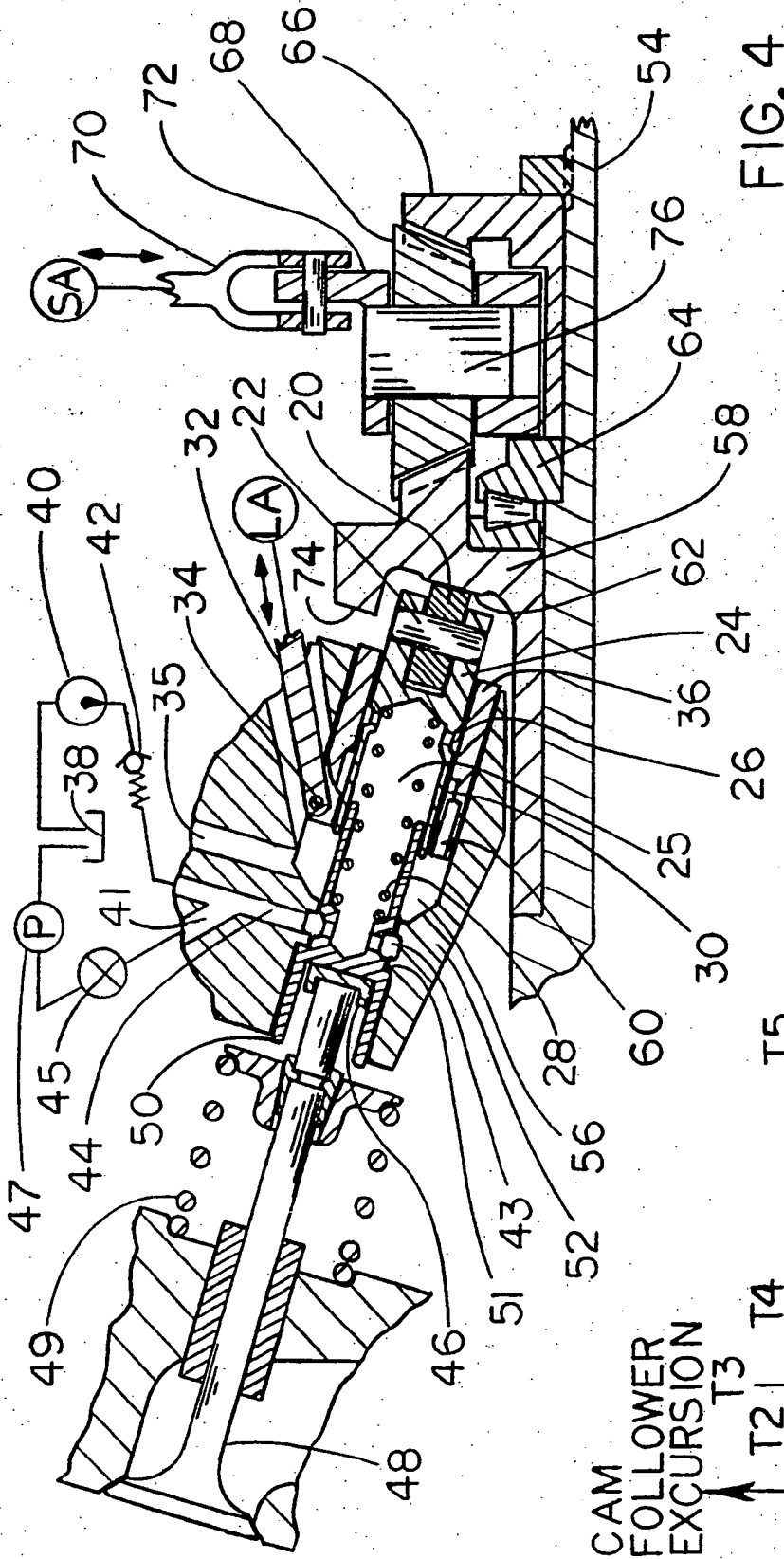


FIG. 4

SUBSTITUTE SHEET

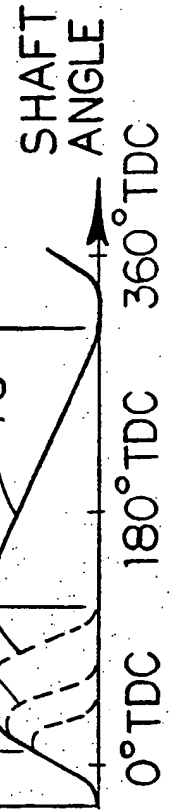


FIG. 5

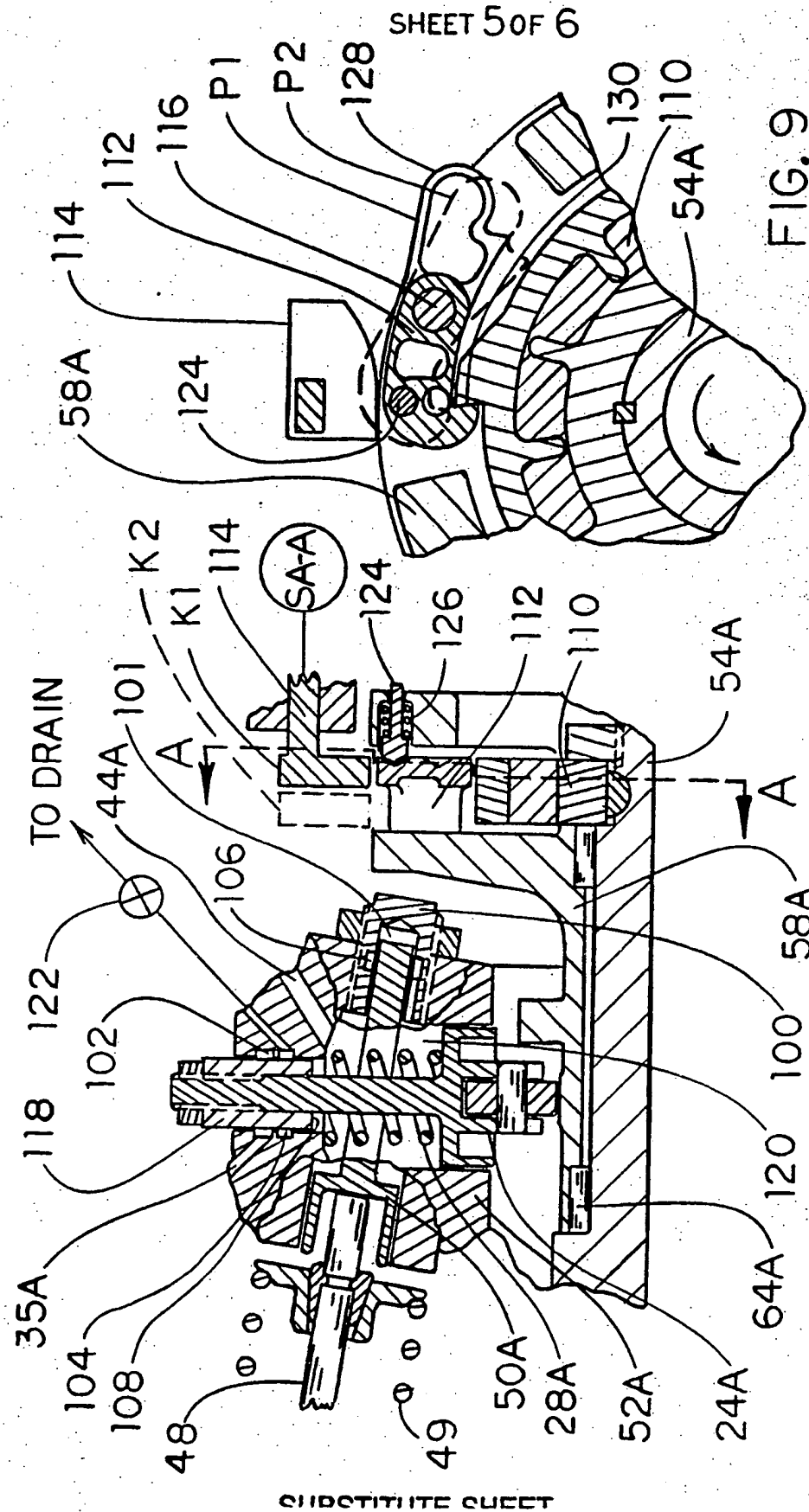
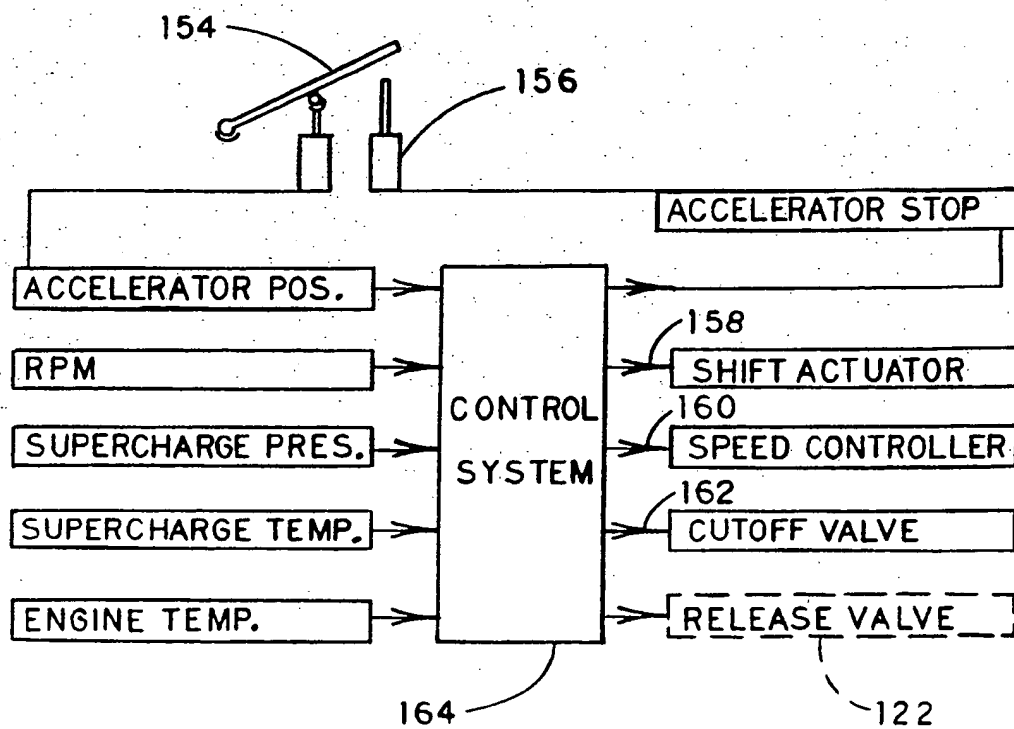


FIG. 9
SECTION A-A
FROM FIG. 8

SHEET 6 OF 6

FIG. 10



SUBSTITUTE SHEET

INTERNATIONAL SEARCH REPORT

International Application No. PCT/US92/00306

| | | | |
|--|--|---|------|
| I. CLASSIFICATION OF SUBJECT MATTER (if several classification symbols apply, indicate all) * | | | |
| According to International Patent Classification (IPC) or to both National Classification and IPC | | | |
| IPC(5): F01L 1/34, F02B 75/22, F02B 75/04 | | US: 123/90.15, 48R, 58A | |
| II. FIELDS SEARCHED | | | |
| Minimum Documentation Searched ⁷ | | | |
| Classification System | Classification Symbols | | |
| U.S. | 123/90.12, 90.15, 90.16, 90.31, 90.32, 90.55, 90.59, 21, 58A, 58AA, 58AB, 48R, 19BF, 64 | | |
| Documentation Searched other than Minimum Documentation to the Extent that such Documents are Included in the Fields Searched ⁸ | | | |
| III. DOCUMENTS CONSIDERED TO BE RELEVANT ⁹ | | | |
| Category ¹⁰ | Citation of Document, ¹¹ with indication, where appropriate, of the relevant passages ¹² | Relevant to Claim No. ¹³ | |
| A | US, A, 4,258,670 (Nohira) See entire document | 29 September 1987 | 3-7 |
| A | US, A, 3,673,991 (Winn) See entire document | 04 July 1972 | 1-31 |
| A | US, A, 4,974,555 (Hoogenboom) See entire document | 04 December 1990 | 1-31 |
| A | US, A, 1,181,463 (La Fontaine) See entire document | 02 May 1916 | 1-31 |
| A | US, A, 1,810,017 (Houston) Figs. 5 and 6 | 16 June 1931 | 1-31 |
| A | US, A, 4,898,128 (Meneely) See entire document | 06 February 1990 | 3-7 |
| A | US, A, 4,892,067 (Paul, et al.) See entire document | 09 January 1990 | 1-31 |
| <p>* Special categories of cited documents: ¹⁰</p> <p>"A" document defining the general state of the art which is not considered to be of particular relevance</p> <p>"E" earlier document but published on or after the international filing date</p> <p>"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)</p> <p>"O" document referring to an oral disclosure, use, exhibition or other means</p> <p>"P" document published prior to the international filing date but later than the priority date claimed</p> <p>"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention</p> <p>"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step</p> <p>"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art.</p> <p>"Δ" document member of the same patent family</p> | | | |
| IV. CERTIFICATION | | | |
| Date of the Actual Completion of the International Search: | | Date of Mailing of this International Search Report | |
| 02 March 1992 | | 03 JUN 1992 | |
| International Searching Authority | | Signature of Authorized Officer | |
| ISA/US | | Thomas N. Moulis | |

FURTHER INFORMATION CONTINUED FROM THE SECOND SHEET

| | | | |
|---|--|-------------------|------|
| A | US, A, 4,168,632 (Fokker) See entire document | 25 September 1979 | 1-31 |
| A | US, A, 4,492,188 (Palmer et al) See entire document | 08 January 1985 | 1-31 |
| A | US, A, 4,510,894 (Williams) See entire document | 16 April 1985 | 1-31 |
| A | US, A, 4,553,508 (Stinebaugh) See entire document | 19 November 1985 | 1-31 |
| A | US, A, 4,153,016 (Hausknecht) See entire document | 08 May 1979 | 3-7 |

V. ☐ OBSERVATIONS WHERE CERTAIN CLAIMS WERE FOUND UNSEARCHABLE¹

This international search report has not been established in respect of certain claims under Article 17(2) (a) for the following reasons:

1. ☐ Claim numbers _____, because they relate to subject matter¹² not required to be searched by this Authority, namely:
2. ☐ Claim numbers _____, because they relate to parts of the international application that do not comply with the prescribed requirements to such an extent that no meaningful international search can be carried out¹³, specifically:
3. ☐ Claim numbers _____, because they are dependent claims not drafted in accordance with the second and third sentences of PCT Rule 6.4(a).

VI. ☐ OBSERVATIONS WHERE UNITY OF INVENTION IS LACKING²

This International Searching Authority found multiple inventions in this international application as follows:

1. ☐ As all required additional search fees were timely paid by the applicant, this international search report covers all searchable claims of the international application.
2. ☐ As only some of the required additional search fees were timely paid by the applicant, this international search report covers only those claims of the international application for which fees were paid, specifically claims:
3. ☐ No required additional search fees were timely paid by the applicant. Consequently, this international search report is restricted to the invention first mentioned in the claims; it is covered by claim numbers:
4. ☐ As all searchable claims could be searched without effort justifying an additional fee, the International Searching Authority did not invite payment of any additional fee.

Remark on Protest

- ☐ The additional search fees were accompanied by applicant's protest.
- ☐ No protest accompanied the payment of additional search fees.

III. DOCUMENTS CONSIDERED TO BE RELEVANT (CONTINUED FROM THE SECOND SHEET)

| Category * | Citation of Document, with indication, where appropriate, of the relevant passages | Relevant to Claim No |
|------------|--|----------------------|
| A | US, A, 4,502,425 (Wride) 05 March 1985 | 3-7 |
| A | US, A, 4,736,715 (Larsen) 12 April 1988 | 1-31 |
| A | US, A, 4,942,853 (Konno) 24 July 1990 | 3-31 |
| A | US, A, 3,356,080 (Howard) 05 December 1967 | 1-31 |